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FORMATION OF DRIVING TORQUE OF ENGINE INTHE TECHNOLOGICAL TRANSPORT MACHINE WITH MULTIPHASE INJECTION OF FUEL

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ABSTRACT

This research analyzes the formation of the torque of the engine of transport and technological vehicles with the mechatronic control system. The influence of multiple injection of diesel fuel with the system "Common Rail" on the unevenness of the engine is considered. Calculations of thermodynamic parameters of the engine working process are performed on the basis of the three-phase fuel injection oscillogram taken using the simulator stand "Car Train Common Rail system". The results of the calculation determine the coefficient of uneven torque for the engine of the prototype with the classic single-phase injection and engine with multiple fuel injections. It is found that the engine with multiple fuel injection coefficients of the unevenness of the torque is close to the value for six-cylinder engines, characterized by perfect balance. With the decrease of the coefficients of uneven torque, conditions of operation of the engine generally improves, the vehicle is better after the wear of their parts due to the weakening of the shock load and vibrations is reduced and as a result, the engine works noiselessly.

***KEYWORDS:** Engine Lug Down and Recovery Coefficient, Common Rail, Multiple Fuel Injection, Irregularity of Torque & The Steadiness of the Engine*

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Original Article

1. INTRODUCTION

For engines of the transport technological machines, operated in the conditions of constantly changing resistance on the working body, the most important indicator of operational characteristic is the engine lug down and recovery coefficient (K_p). The size of this coefficient is defined by the relation of the maximum driving torque by the time of, developed by the engine on the nominal rotary speed of the bent shaft. The size K_p, is higher, the more loading overcomes the engine without gear shift in the gear box. At the same time, the range of change of rotary speed of bent shaft in which the engine works steadily is important. The more this range, the operation of the engine at the changing loadings is steadier (Shtain, 2015). On Figure 1, operational characteristics of engines of the construction and road equipment of outdated type with a mechanical drive gear of steering of the fuel equipment of high pressure are presented.

By the operational characteristic of the YaMZ-238DE engine offset the size of the maximum driving torque towards the smallest rotation rate (RPM) – 1300 min^{-1} contrary to (1500…1600) min^{-1} at the basic engine YAMZ-238. The amount of engine lug down and recovery coefficient for YaMZ-238DEwill be equal toK_p=$M_{e.max}$/$M_{e.nom}$=1274/1100 =1.15. Meanwhile, the stability zone of RPM at the varying loads is (1300…2100) min^{-1}, that is a satisfactory factor. The amount of engine lug down and recovery coefficient for Komatsu-SAA6D125EequalsK_p=1580/1300=1.21, stability zone of RPM is – (1300…1900) min^{-1}.

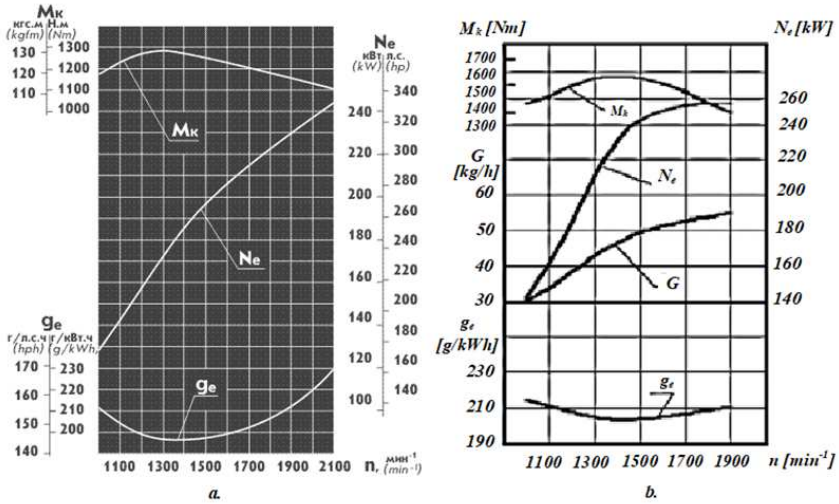

Figure 1: Operational Characteristics of Engines: a) YaMZ-238DE; b) Komatsu-SAA6D125E

The optimal amount of engine lug down and recovery coefficient is achieved due to increasing of rapid curve of driving torque in the work area of engine performance. As shown in the Figure 1b, a good result of engine lug down and recovery coefficient is achieved by more flat smooth curve of driving torque, in the work area of engine performance due to its derating. In the Figure 2, operational characteristics of new engines YaMZ-650.10 иMAN-D2865LF21, which are used for transport and technological vehicles with mechatronics control system and multiple fuel injection, are shown.

The major factors influencing the size of driving torque define the following dependence:

$$M_e = N \cdot \frac{\eta_v \cdot p_k}{\alpha} \tag{1}$$

Where, N– coefficient of proportionality, η_v – coefficient of charge, $p_к$ – charging pressure, α – excess-air coefficient.

Changing the specified factors, it is possible to achieve the formation of the curve of driving torque in the corresponding zone of loadings and RPM (Shtain & Panfilov, 2018).

Influencing on fuel injection rate, it is possible to provide the required nature of change α. Reduction limits of $\alpha$are connected with the standards of smoking at the exhaust, therefore possibilities of increase in the engine lug down and recovery coefficient without pressure charging are limited, and it usually does not exceed 10 … 15%. In case of usage of pressure charging control in the engine, it becomes possible to influence the value of driving torque by the setting $p_к$. The experience of exploitation, experience in the operation of pressure charging engines shows that, the engine lug down and recovery coefficient can be brought to a value of 1.3...1.5. Coefficient of engine charge is determined primarily by the installation of the valve timing setting (it is desirable that they are adjustable at the expense of mechatronic control systems). In the analyzed engines, the valve timing can be customized.

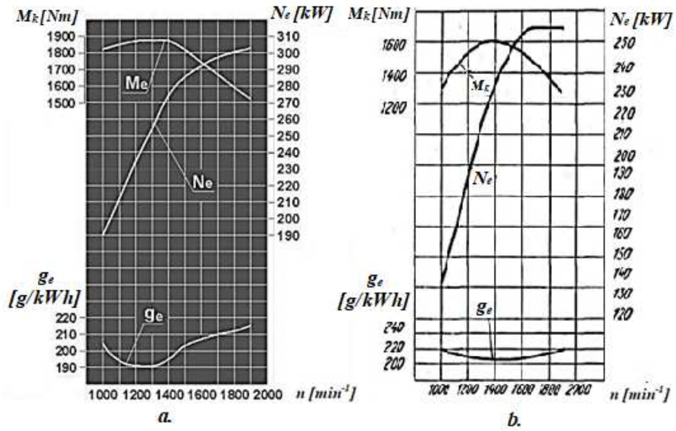

Figure2: Operational Characteristics of Engines: a) YaMZ -650.10; b) MAN-D2865LF

In modern engines of transport and technological machines in the evaluation of performance, there are definitions such as areas of constant power and torque. Due to the area of constant power in the working area of the engine increases the steepness of the torque, and thus the coefficient of adaptability. Due to the area of the torque, the area of stable operation of the engine under overload conditions is expanded and, as a result, there is no need to change gears in the transmission units. In the engine YAMZ-650.10 (Figure 2. a), these areas are clearly visible and the value of the coefficient of flexibility is equal to $K_p=M_{e.мах}/M_{e.n}$=1200/1100=1,09, i. e. slightly less than that of theYaMZ-238DE engine. At the same time, the area of stable operation has been significantly expanded (1000…2000) min^{-1}. In the engine MAN-D2865LF2 (Figure 2b), there is only the area of the constant power in the working area, meanwhile the value of the coefficient of flexibility is equal to $K_p=M_{e.мах}/M_{e.n}$=1600/1250=1.28, the area of stable operation of the engine will be (1400…1900) min^{-1}, i. e. less than that of the YaMZ-650.10 engine.

Therefore, to form the required value of the coefficient of engine, lugdown and recovery is necessary:

- To modify the fuel injection rate during the lugging of the engine;
- To modify the charging pressure at the reduction in the frequency of rotation of the bent shaft;
- To modify phases of the valve timing setting at the changing of the frequency of rotation of the bent shaft.

It is of interest, how the driving torque is formed with multiple fuel injection within the working cycle on the example of a specific brand of engine. At present, not only economic but also environmental performance of engines is improving to meet the increasingly stringent requirements for harmful emissions and noise of vehicles regulated by the government. For vehicles` engines, the most important thigs are the minimal specific fuel consumption, emissions of nitrogen and carbon oxides, polycyclic aromatic hydrocarbons (and for diesel engines — solid particles), and the permissible noise level. Drop of blowouts of harmful substances internal combustion engines can be reached: impact on working process (creation of mixtures and combustion) and engine design; neutralization of toxic components in the final engine system; application of alternative (more environmentally friendly) fuel; providing normal service conditions of the vehicle (modes of the movement, technical condition of the engine, quality of fuel, etc.). As a rule, simultaneous performance of the rigid ecological and improved fuel and economic indicators of the engine is difficult realizable. It is known that in most cases, the requirements of environmental standards are fulfilled, mainly by the de-rated engine, application of recirculation of the fulfilled gases, etc., and it involves deterioration in the heat usage of the working process and according to fuel efficiency.

DETERMINATION OF PARAMETERS OF MULTIPLE FUEL INJECTION

In modern designs of diesel with the accumulator system of fuel feeding, "Common Rail" multiple fuel injection is applied. In Figure 3 there is the oscillogram of three-phase fuel injection, which is made with the usage of «Car Train Common Rail system». It is considered that multiple fuel injection allows reducing ignition delay period, connected with the fact that injection of the main portion of the fuel is carried out in the area, which is in some extent homogenized and ionized by development of cold and ardent reactions of oxidation of ignition portion of the fuel. As a result of the speed of allocation of warmth in the initial phase of combustion, it considerably decreases in comparison with single-phase (traditional) injection. At such, transformation of combustion process (in initial phase) the rigidity, the maximal pressure and temperature of combustion decreases and it allows to use fuel with lower cetane number, and, also, to reach drop of thermal and mechanical loading, noisiness of the engine, level of harm and smoke of the fulfilled gases. It is known from the theory of combustion process in heat engines (Lukanin, 2007) that suppression of speed of allocation of warmth in the initial phase of combustion around the top dead point (UDP), where warmth will be transformed to work most effectively that creates prerequisites to the drop of heat usage and by that profitability of cycle. It is necessary to pay attention (Figure 3) that injection of the main phase fuel begins after UDP, it is typical for modern diesels with the system «Common Rail», where the fuel with the high cetane number is used and there is no need to have an advance angle of fuel injection.

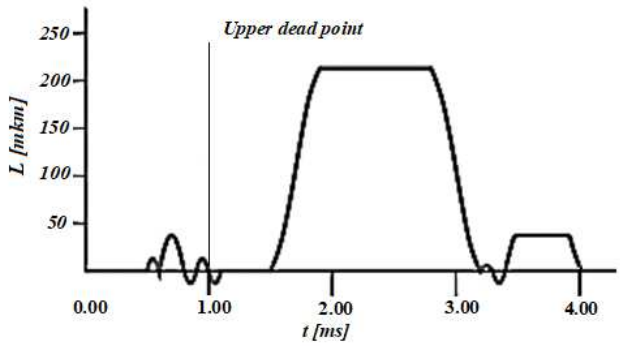

Figure3: Oscillogram of the Three-Phase Fuel Injection

Preliminary injection prepares the combustion chamber for the main injection. Thanks to the small amount of previously injected fuel, there is prevention typical for diesel engines of rigid work as the combustion procedure occurs more softly. At preliminary injection, before the process of the main injection, small amount of diesel fuel is entered into the cylinder, the combustion chamber is prepared for the realization of the main injection. On the oscillogram, it is displayed in the form of a typical, very short electric impulse of the solenoid valve. Between stages of preliminary and main injection, on the magnetic valve in injector current supply does not appear. In conclusion, there is an injection of the main quantity of fuel corresponding to the relevant engine operation mode. In stages of the main injection, the needle lift is much more, than in stages of preliminary and final injection. Thanks to the final injection, the temperature of the fulfilled gases increases. This effect is used for regeneration of the system of neutralization of the fulfilled gases in the black filter and in the accumulative catalyst. It is known from the theory of thermodynamic cycles of heat engines (Kirilin, 1973) that the greatest effect of the heat usage of the working body is provided at fast fuel combustion on the line of an expansion near TDC, i. e. at the realization of the cycle of N. Otto (Figure 4).

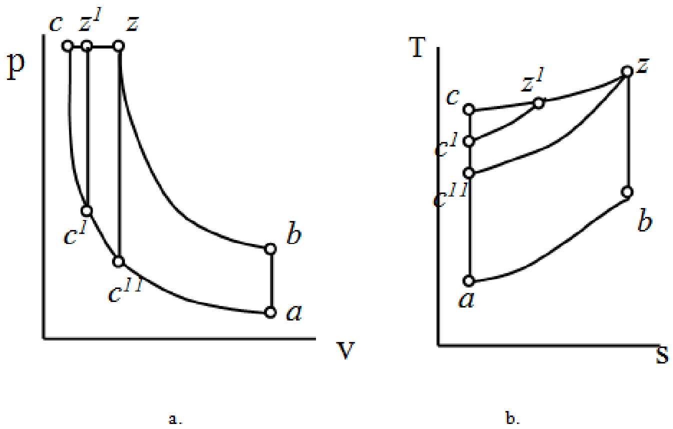

Figure4: Comparative diagram of Cycles of Thermal Engines with different Methods of Heat Supply: a - in the Coordinates p–V (Pressure-Volume of the Working Fluid in the Cylinder); b – in the Coordinates T –S (Temperature-Entropy of the Working Fluid in the Cylinder)

The aspiration to reduce the size fuel injection rate near TDC (so-called "sub injection") and giving of fuel at the end of expansion process (subsequent "injection") reduces the effect of transformation of warmth to work. As you can see in the diagram (Figure4), the highest amount of the thermal efficiency of cycles (η_t) with the same compression ratio is provided at cycle with warmth supply at the constant volume: a-c^{11}-z-b.

$$\eta_t = 1 - \frac{T_z}{T_b} \tag{2}$$

The subsequent fuel injection at the end of the expansion process increases temperature T_b and by that, in addition, reduces amount η_t. For quantitative assessment of efficiency of the organization of the working process in the engine with warmth supply at the constant volume, we will use the formula, known in thermodynamics:

$$\eta_t = 1 - \frac{1}{\varepsilon^{k-1}} \cdot \frac{\lambda \cdot \rho^k - 1}{(\lambda - 1) + k\lambda(\rho - 1)} \tag{3}$$

Where, ε- compression ratio, in calculations for the engine from which the oscillogram was taken, the accepted amount -16, 6; *K*– indicator of the adiabatic curve of the working body – 1, 4; λ – degree of pressure boost of the working body during combustion (depends on engine loading, the accepted amountis2, 4); ρ–the extent of expansion of the working body during combustion (depends on the duration of fuel injection).

For the working process of the engine with single-phase injection of fuel size, "ρ" was accepted as equal to 1.5 (at traditional diesels with regular set-up of the fuel equipment of high pressure), for multiple fuel injection - respectively 2 and 2.5. Results of calculation for the determination of the expected thermal efficiency are presented in Figure 5.

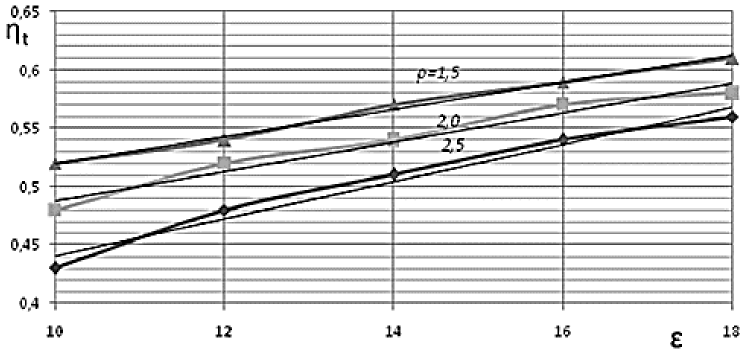

Figure 5: Diagram of Change of Thermal Efficiency (η_t) from the Values of ε and ρ

On the diagram, it is visible that almost always the amount of η_t depends upon the amount of ρ. If ρ= 1,5, the amount of η_t is 0,60, if ρ=2; 2,5, the amount of η_t is 0,57…0,55, respectively.

Therefore, at the level of heat usage of the injected fuel in loss engine cylinder on efficiency, the fuel efficiency will make up to 10%. Obviously, the application of multiple fuel injection in the engine solves the problem on performance of rigid environmental standards for the automobile vehicles used in the cities and settlements.

In this work, on the basis of the oscillogram of three-phase injection of fuel (Figure 3), removed with the use of «the Car Train Common Railsystem» stand exercise machine calculations for the definition of fuel injection rate of the corresponding certain brand of the engine are executed.

Time of the fuel expiration by the spray jet at the main injection is (3.3-1.5)=1.8ms (as it can be seen on the oscillogram), that makes the amount of the crank angle (we will accept the nomial rotary speed of the standard diesel transport and technological vehiclesapproximately equal (2000 min^{-1})) equal:$\varphi=t\cdot 6\cdot n$=1.8$\cdot 10^{-3}\cdot$ 6$\cdot$ 2000=21.6φ^0**.**

The average speed of the expiration of fuel (м/с) throughnozzle bores of spray jet is calculated by the formula:

$$\omega_n = \sqrt{\left(2/\rho_f\right)\cdot\left(p_n - p_v\right)} \tag{4}$$

Where, p_n — medium pressure fuel injection, Pa; $p_v=(p_c+p_z)/2$ - medium pressure of the gas in the cylinder at the moment of injection; p_cиp_z - the pressure at the end of compression and the combustion determined by data of thermal calculation of the engine.

We will accept the amout of pn for the fuel system «Common Rail» equal to 150$\cdot$106 Pa, the amout ofp_v, for the majority of diesels equal to9$\cdot 10^6$ Pa, firmness of fuel p_f=850 кг/м^3, then the average speed of fuel expiration by the spray jet will be equal to575 m/sec. The total area of nozzle bores of spray jet (f_c) will be: (the accepted number of nozzle bore-5, the diameter –(0.2 mm), the amount – 0.157 mm^2. Fuel injection rate (mm^3)will be calculated by the formula:

$$v_c = f_c\cdot\mu_n\cdot w_f\ t\cdot 10^3 \tag{5}$$

Where, μ_n – fuel expense coefficient through injector cones, the accepted amount is about 0.65-0.85. Substituting the received values, we will receive V_c=105 mm^3.

Results of calculation of fuel injection rate in thespray jet are presented on theFigure6.

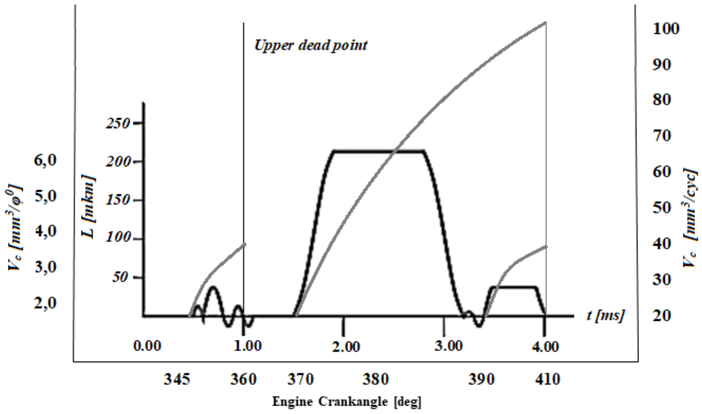

Figure 6: Characteristics of Multiple Fuel Injection

Knowing the amount of fuel injection rate on the nominal mode of the engine, it is possible to determine its rated power from the formula:

$$V_c = \frac{g_e \cdot P_e \cdot 10^3}{30 \cdot n \cdot i \cdot \rho_f} \tag{6}$$

Where, g_e- specific effective expense of fuel; P_e- effective rated power of the engine; n – nominal rotary speed of the bent shaft of the engine; i – number of cylinders of the engine; ρ_f- firmness of fuel.

The received power size on the nominal mode is closeto (P_e=*100kW*) for the modern engine YaMZ-53441.

THERMODYNAMIC CALCULATION

The author's CAD-SAPR program has been used for the creation of an indicator diagram of the YaMZ-53441engine and the subsequent kinematicanddynamiccalculations. Thermal calculation of this engine was carried out on the basis of the specification, thermodynamic parameters of the working process of the engine by results of the calculation are determined.

By results of thermodynamic calculation, the developed indicator chart of the YaMZ-53441 engine is constructed of the condition of single-phase injection of fuel (Figure7).

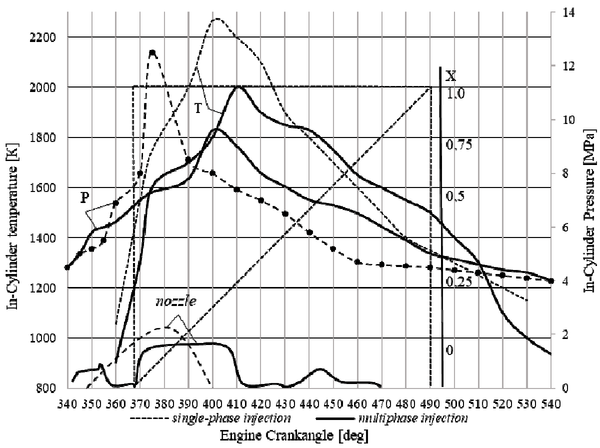

Figure 7: The Results of the Calculation and Plotting of the Indicator Diagram of the Engine YaMZ-53441

To create the developed indicator diagram of the engine with multiple fuel injection, taking into account that two phases (Figure6) at the beginning have an impact on the nature of the process of expansion, we will calculate combustion procedure temperatures in characteristic points and then, using thermodynamic dependences, we will define values of pressure of working body in the corresponding points. Forthat, in the basic data, we will accept efficiency of warmth coefficient during combustion (ξ) equal to 0.8 in order to get 10% reduction of indicator efficiency (η_i) for engines with multiple fuel injection (Shtain & Panfilov, 2015).

The gas temperature is determined by the constant formula:

$$T = T_a \cdot \frac{p}{p_a} \cdot \frac{V}{V_a} \cdot \frac{1}{\mu_x} \tag{7}$$

Where, T_a -the charge temperature at the beginning of compression calculated by the formula:

$$\mathrm{T}_a = \frac{T_0 + \Delta T + \gamma_r \cdot T_r}{1 + \gamma_r} \tag{8}$$

Where, p- the pressure which is taken off of the indicator diagram;p_a- the pressure at the beginning of the compression process, the accepted amount is 0.139МПа; V_a- cylinder capacity (1.17л);V- the current amount of cylinder capacity;

$$V = \frac{V_h}{\varepsilon - 1} + 0{,}785 D^2 \cdot \frac{S}{2}(1 - \cos\varphi - \frac{\lambda}{2} sin^2\varphi) \tag{9}$$

ΔT- the amount of heating up of fresh charge from cylinder walls, the accepted amount is 5^0; *Tr*- temperature of residual gases, the accepted amount is750 К; *γr*- coefficient of residual gases, the accepted amount is 0,033; V_h- cylinder capacity (1,1liters); Ɛ- compression ratio (17, 5); *D*- cylinder bore (105 mm); *S*- piston stroke (128mm); *λ*- crank radius relation to connecting rod length *L* (0, 27).

The coefficient of molecular change of mixture during combustion is determined at any moment by the formula:

$$\mu_x = 1 + \frac{\mu_0 - 1}{1 + \gamma_r} \cdot x \tag{10}$$

Where, μ_0 - theoretical coefficient of molecular change of the mixture:

$$\mu_0 = 1 + \frac{0{,}0639}{\alpha} \tag{11}$$

Where,α - the coefficient of the excess of air at the nominal mode is accepted by equal 1.8; *x* –the share of the burned-down fuel during the process of combustion.

Therefore, mainparameters of the engine with single-phase injection on temperature are:

the temperature of the end of the compression T_c=1056 К, the highesttemperature of combustion procedure T_z=2267К, withmultiple fuel injection respectively: 903 Кand 1995К.

Results

To define the driving torque forming on the crank of one cylinder with use of the CAD program, it is executed kinematic and dynamic calculations.

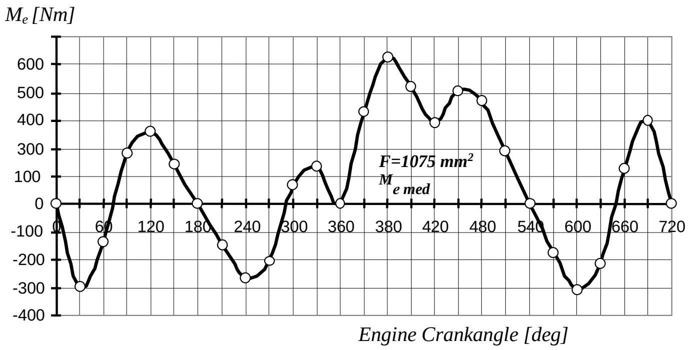

Figure 8: Formation of Engine Torque with Single-Phase Fuel Injection

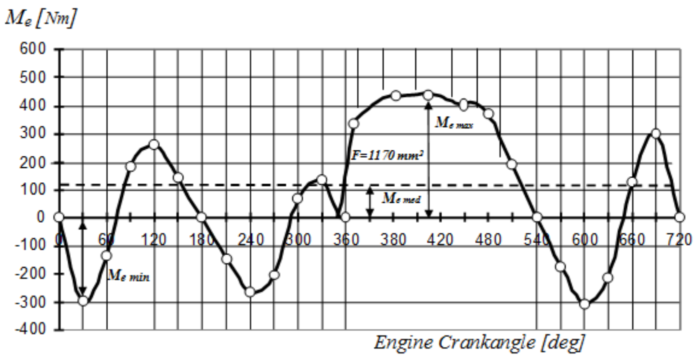

Figure 9: The Formation Torque of the Engine with Multiple Fuel Injection

In the system engine-transmission, the driving torque - M_e due to its unevenness on crank angle (Figure 9) causes variable turning of elements of transmission (shifting jerks), and also variable reactions on engine bearers. The unevenness of driving torque is estimated by unevenness coefficient.

$$\mu = \frac{M_{e\max} - M_{e\min}}{M_{emed}} \tag{12}$$

By calculating the size for the engine pilot model with classical single-phase injection (Figure8), and the YAMZ-53441 engine with multiple fuel injection (Figure 9), the following results respectively are received:

$$\mu_{s.p.} = \frac{620 - 270}{100} = 3.5$$

$$\mu_{m.p.} = \frac{440 - 270}{115} = 1.5$$

The amount of μ varies depending on the mode of operation of the engine; at the same time, the change in the speed mode, where inertial forces prevail affects the numerator, the load mode affects the denominator of the above expression.

CONCLUSIONS

Theoretical and experimental (operational) results show that the overall operation of the engine of the transport and processing machine as a whole has an even greater effect on the engine than its balance. The amount of $\mu_{m.p.}$=1,5 approaches the value of the coefficient of unevenness for six-cylinder engines, characterized by perfect balance. With an increase of the uniformity of torque (at the decrease of $\boldsymbol{\mu}$), the operating conditions of the engine and machinery and components of the machine as a whole improve significantly, the vehicle moves off better, the wear of their parts due to the weakening of the shock load and vibrations associated with uneven engine running decreases, and as a result, the engine running silently.

It should also be noted that, according to the results of the area calculation ($F=1170\ mm^2$) generated by the torque during the expansion cycle of the engine with multiple fuel injection in comparison with the engine with single-phase fuel injection ($F=1075\ mm^2$), an increase in useful indicator works and according to the average torque.

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